Design of Shipping Containers for Master Equatorials

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The delicate nature of the Master Equatorials makes them highly susceptible to damage, especially during transit. A special container has been designed and built for the purpose of shipping assembled Master Equatorials to overseas antenna sites in Australia and Spain. The design features of the shipping containers are outlined in this article, and the advantageous use of motor vehicle shock absorbers is described.

I. Introduction

Because of the delicate nature of the assembled Master Equatorials, a special container was designed and built for the purpose of shipping them from JPL to their overseas antenna sites in Australia and Spain. (Shipment of the units in the disassembled state was considered and rejected because of the expense of constructing adequate reassembly areas near the antennas.) The special ball bearings on both the declination and polar axles are most delicate from a damage point of view; however, the worm gear teeth and the main mirror are also items of concern.

A consideration of possible damage to the ball bearing races caused by acceleration led to the establishment of a 5-*g* maximum vertical acceleration. Discussions with transportation experts led to the criterion of the ability to sustain a 0.30-m (12-in.) vertical drop of the container without allowing the Master Equatorial to exceed 5-*g* acceleration. Horizontal accelerations were limited to a value of 1 *g*.

Prevention of fretting corrosion of the bearing races and balls was also studied. It is recognized that during transit in both airplanes and ground vehicles a steady-state high-frequency excitation may exist for long periods of time. An examination of statistical studies (Ref. 1) suggested that sufficient isolation from such vibration could be obtained by making the natural undamped frequency of the packaged Master Equatorial not more than 3.5 Hz. During shipment both axles of the Master Equatorial are clamped tightly to prevent any oscillation. Both worms are backed away from mesh with their worm gears so as to prevent any possible damage.

II. Description of the Package

The main elements of the package are shown schematically in Fig. 1. The mass of the Master Equatorial is approximately 1820 kg and the suspended base mass is 360 kg. The center of gravity of the total suspended mass of 2180 kg is approximately 0.83 m above the Master Equatorial base and is equidistant from the four springs. The suspended base is a weldment made of rectangular steel tubing $0.103 \times 0.254 \times 0.0048$-m wall thickness. Each spring is actually a cluster of six die springs in parallel. A hydraulic shock absorber is mounted within each spring cluster. Four radius rods connect the suspended weldment to the main base, which is of laminated wood construction with a steel angle border. The Master Equatorial is wrapped in a vapor barrier, and its base is bolted to the suspended base. The springs and shock absorbers are mounted so that they can resist either upward or downward acceleration forces.

The cover is made of aluminum alloy sheet and is bolted to the main base. The overall dimensions of the package are 1.53 × 2.08 × 2.26 m high; its mass, including the Master Equatorial, is approximately 2600 kg. The lower part of the cover can be disassembled and stored within the upper part which when attached to the main base gives an overall height of 1.02 m. Since only one container was built for shipping two Master Equatorials, the reduced height facilitates the return shipment of the container.

The detail design is shown on JPL Drawing 9350283. Figures 2 and 3 show various design details. The container was built by the Boller and Chivens Division of the Perkin Elmer Corporation.

III. Dynamic Analysis

Since the mass of the suspended parts is approximately 85% of the total mass of the package, and because the unsprung base is largely wood, it is believed that a simple one-degree-of-freedom model will approximate the behavior of this system. The elements of this system are shown in Fig. 4.

The differential equation of motion of this system is

$$m\ddot{x} = -kx - b\dot{x} + mg \tag{1}$$

subject to the initial conditions,

$$x_0 = 0, \qquad \dot{x}_0 = \sqrt{2gh} \tag{1a}$$

where

m = mass

k = total spring constant in force per length

b = damper coefficient in force per unit velocity

g = acceleration of gravity

x_0 = initial displacement

$\dot{x}_0$ = initial velocity

$\dot{x}$ = velocity

$\ddot{x}$ = acceleration

and let

$\omega^2 = \dfrac{k}{m}$ be the undamped natural frequency

$\zeta = \dfrac{b}{2m\omega}$ be the damping ratio

$q = \omega\sqrt{1-\zeta^2}$

The solution of Eqs. (1) and (1a) is

$$x = e^{-\zeta\omega t}\left[-\frac{g}{\omega^2}\cos qt + \left(\frac{\sqrt{\dfrac{2gh}{\omega^2}} - \dfrac{\zeta g}{\omega^2}}{\sqrt{1-\zeta^2}}\right)\sin qt\right] + \frac{g}{\omega^2} \tag{2}$$

The first three time derivatives of Eq. (2) are, respectively,

$$\dot{x} = e^{-\zeta\omega t}\left[\sqrt{2gh}\cos qt + \left(\frac{\dfrac{g}{\omega} - \sqrt{2gh}\,\zeta}{\sqrt{1-\zeta^2}}\right)\sin qt\right] \tag{3}$$

$$\ddot{x} = e^{-\zeta\omega t}\left[(g - 2\zeta\omega\sqrt{2gh})\cos qt + \left(\frac{\sqrt{2gh}\,\omega\,(2\zeta^2-1) - \zeta g}{\sqrt{1-\zeta^2}}\right)\sin qt\right] \tag{4}$$

$$\frac{d\ddot{x}}{dt} = e^{-\zeta\omega t}\left\{[-2\omega\zeta g + \omega^2\sqrt{2gh}\,(4\zeta^2-1)]\cos qt + \left[\frac{\zeta\omega^2\sqrt{2gh}\,(3-4\zeta^2) + \omega g\,(2\zeta^2-1)}{\sqrt{1-\zeta^2}}\right]\sin qt\right\} \tag{5}$$

For the problem at hand, the important quantities are the maximum absolute values of $\ddot{x}$ and x, because they determine, respectively, the maximum force on the suspended mass, and the necessary clearance between the suspended mass and the container. The maximum displacement $x_{\max}$ can be found by setting the bracketed term of Eq. (3) equal to zero, determining qt, and substituting its value into Eq. (2). The maximum value of $\ddot{x}$ occurs at $t = 0$ if Eq. (5) is not negative for $t = 0$. Otherwise the maximum value of $\ddot{x}$ is found by setting the braced term of Eq. (5) equal to zero, determining qt, and substituting its value into Eq. (4). These results are

$$x_{\max} = e^{-\zeta\omega t}\left[\frac{1}{\omega^2}\sqrt{g\,(2h\omega^2 + g - 2\sqrt{2gh}\,\omega\zeta}\right] + \frac{g}{\omega^2} \tag{6}$$

where

$$\zeta\omega t = \frac{\zeta}{\sqrt{1-\zeta^2}}\left[\arctan\frac{\sqrt{1-\zeta^2}\sqrt{2gh}}{\sqrt{2gh}\,\zeta - \frac{g}{\omega}}\right] \tag{6a}$$

and

$$\ddot{x}_{\max} = g - 2\zeta\omega\sqrt{2gh} \tag{7}$$

when

$$[-2\omega\zeta g + \omega^2\sqrt{2gh}\,(4\zeta^2 - 1)] \geq 0$$

$$\ddot{x}_{\max} = -e^{-\zeta\omega t}\sqrt{2\omega^2 gh + g^2 - 2\omega g\zeta\sqrt{2gh}} \tag{7a}$$

where

$$\zeta\omega t =$$

$$\frac{\zeta}{\sqrt{1-\zeta^2}}\left\{\arctan\frac{\sqrt{1-\zeta^2}\,[\omega^2\sqrt{2gh}\,(1-4\zeta^2) + 2\omega g\zeta]}{-\omega g\,(1-2\zeta^2) + \omega^2\sqrt{2gh}\,(3\zeta - 4\zeta^3)}\right\} \tag{7b}$$

when

$$[-2\omega\zeta g + \omega^2\sqrt{2gh}\,(4\zeta^2 - 1)] < 0$$

Thus, the important quantities $x_{\max}$ and $\ddot{x}_{mar}$ are given in terms of the parameters stipulated in the design specification and the damping ratio.

Displacements and accelerations were computed using the following parametric values:

$\omega = 19.4$ radians per second

$h = 0.305$ meter

$g = 9.8$ meters per second squared

Figure 5 gives the displacement ratio x/x_{ST}, where x_{ST} is the static displacement, and the acceleration in g as a function of time for three different damping ratios. Figure 6 shows the maximum displacement ratio and the maximum acceleration as a function of the damping ratio. Although the original specification of 5 g was met with zero damping, the corresponding displacement was difficult to accomplish in the design. As Fig. 6 demonstrates, a damping ratio of 0.40 reduces the maximum displacement ratio from 6 to 3.5, and reduces the acceleration from 5 to 3.6 g.

IV. Selection of the Shock Absorbers

The preceding equations are based upon a viscous damper; that is, one whose resisting force is proportional to the velocity. Commercial hydraulic shock absorbers are likely to vary considerably from this characteristic, which is probably why manufacturers are reluctant to specify a force per unit velocity. Reference 2 states that dampers involving orifice flow have forces more nearly proportional to the square of the velocity.

Two different motor vehicle shock absorbers were tested. The more expensive one had entirely different characteristics in compression and extension, whereas the other one, which cost \$6.00, appeared to be the same in the two modes. The shock absorber was clamped in a vise and moved by exerting a constant force through a spring balance scale. The time intervals to move various distances under a constant force were recorded. The damper coefficient b was computed as the product of force and time divided by distance. Different values of force were used without affecting the computed value of b. However, all forces were small in comparison to what would occur at the end of the specified drop for the container, this latter force being 100 to 200 times the test forces. As first tested, the shock absorber had b values of 1750 to 2600 newtons per meter per second. The hydraulic fluid was drained and replaced with SAE No. 30 engine oil, with the result that the b value increased to 8400 and 10,500 newtons per meter per second, respectively, for compression and extension. Considering that there are four absorbers in parallel, the damper ratio was computed to be 0.40 in compression. These are the units which are presently installed in the shipping container.

The addition of four very inexpensive commercial parts has reduced considerably both the maximum acceleration and displacement as calculated by the foregoing analysis. Drop tests were not made on the shipping container, but it has been used successfully in transporting the Master Equatorials to their overseas sites, since tests on the installed instruments have verified that there has been no damage incurred.

References

1. *Shock and Vibration Handbook,* Vol. 3, Chap. 47, Edited by C. M. Harris and C. E. Crede, McGraw-Hill Book Co., Inc., New York, 1961.

2. *Shock and Vibration Handbook,* Vol. 2, p. 31-12, edited by C. M. Harris and C. E. Crede, McGraw-Hill Book Co., Inc., New York, 1961.

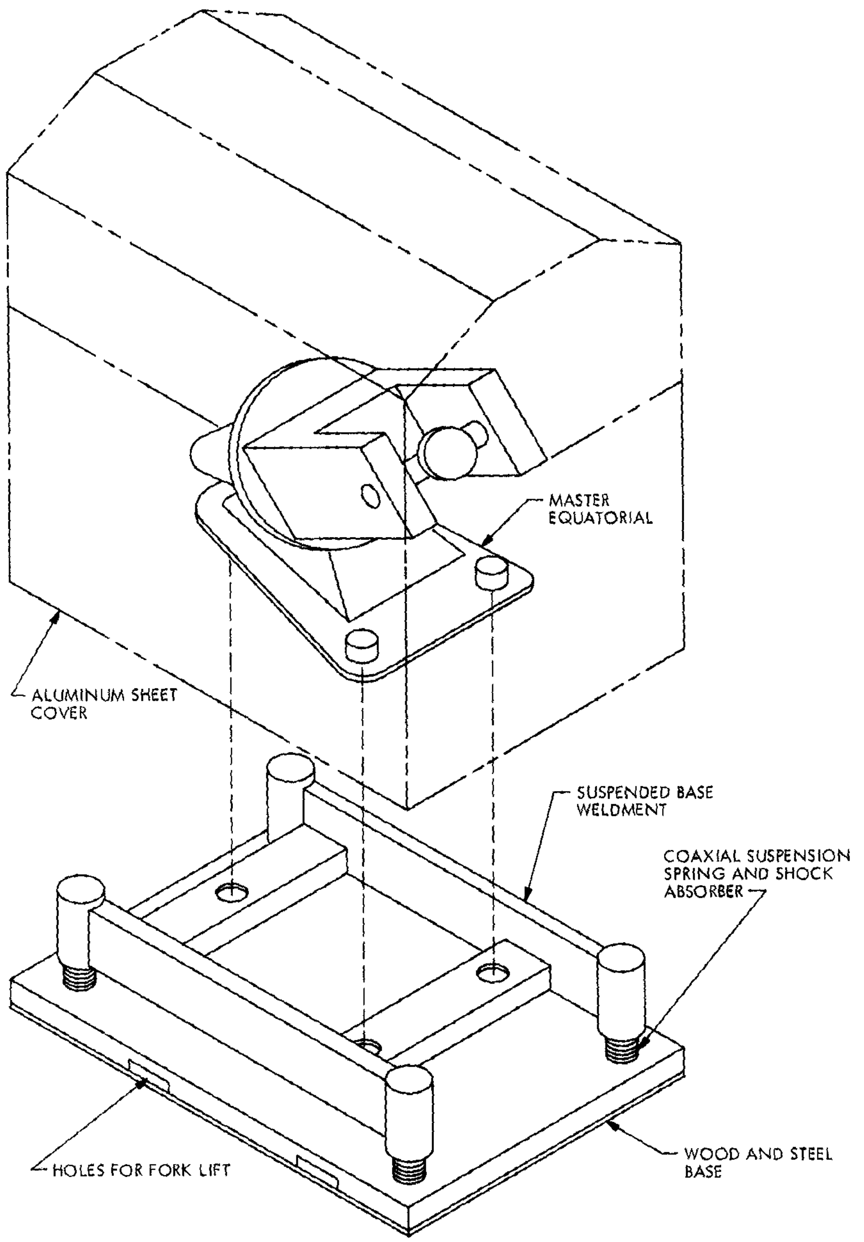

Fig. 1. Exploded view of packaged Master Equatorial

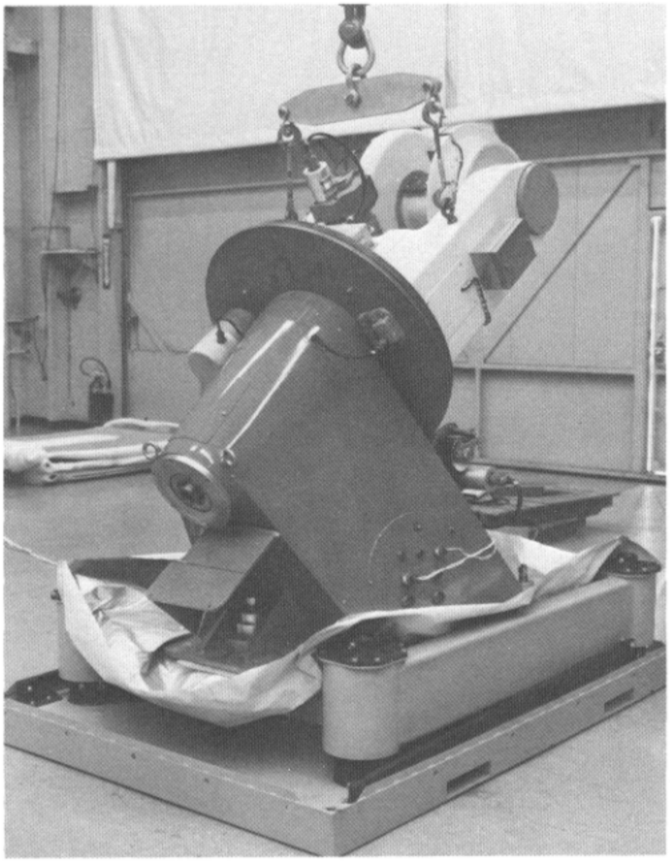

Fig. 2. Master Equatorial being placed on container base

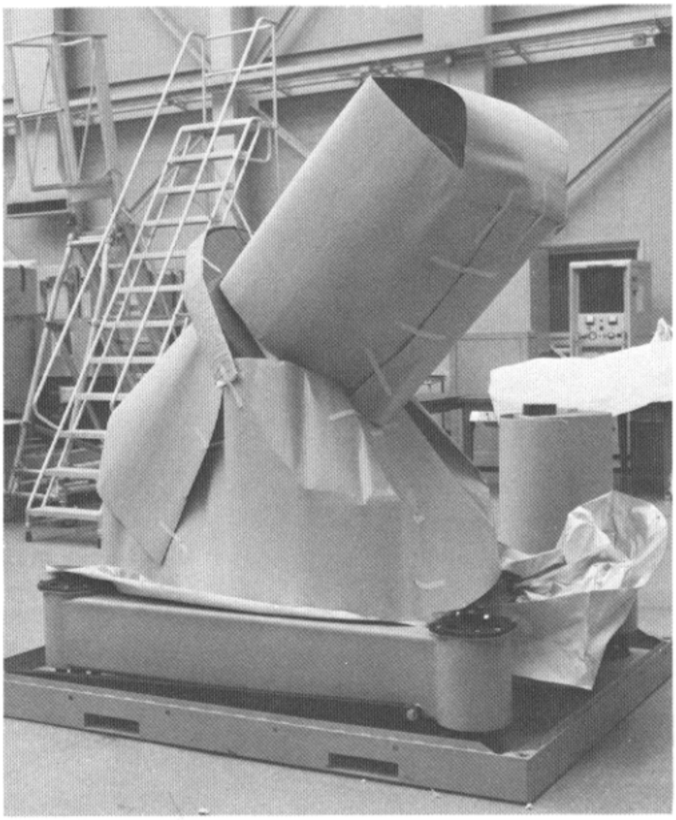

Fig. 3. Preparation of Master Equatorial

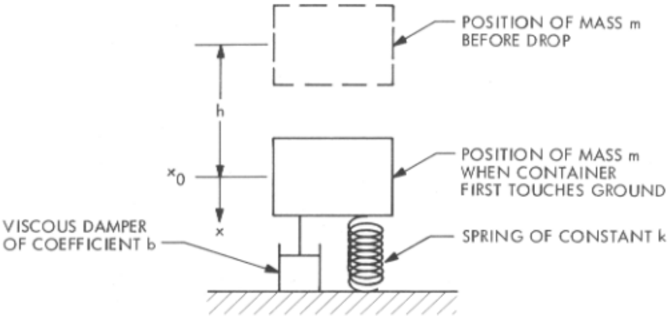

Fig. 4. Dynamic model

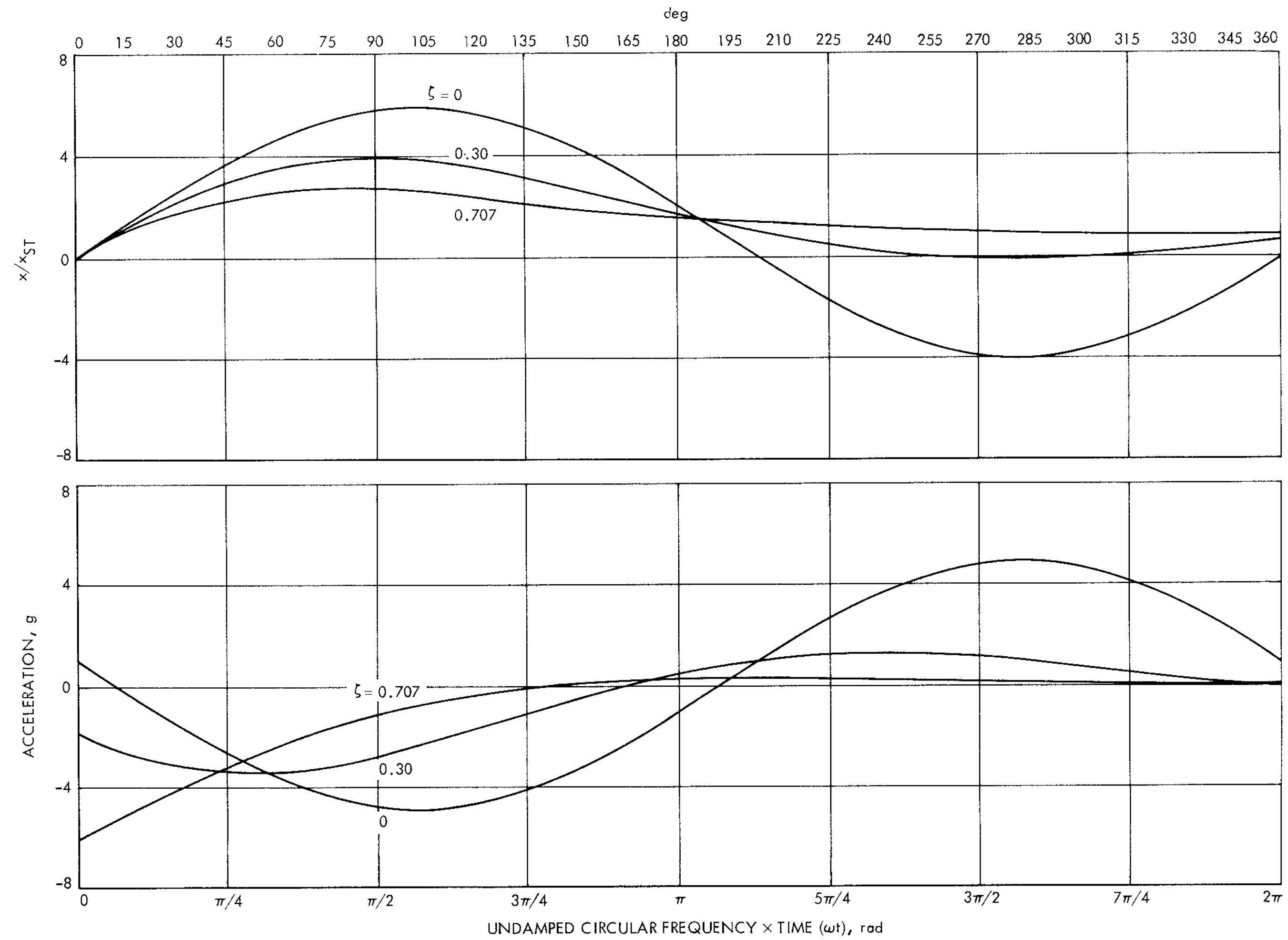

Fig. 5. Displacement and acceleration versus angular displacement

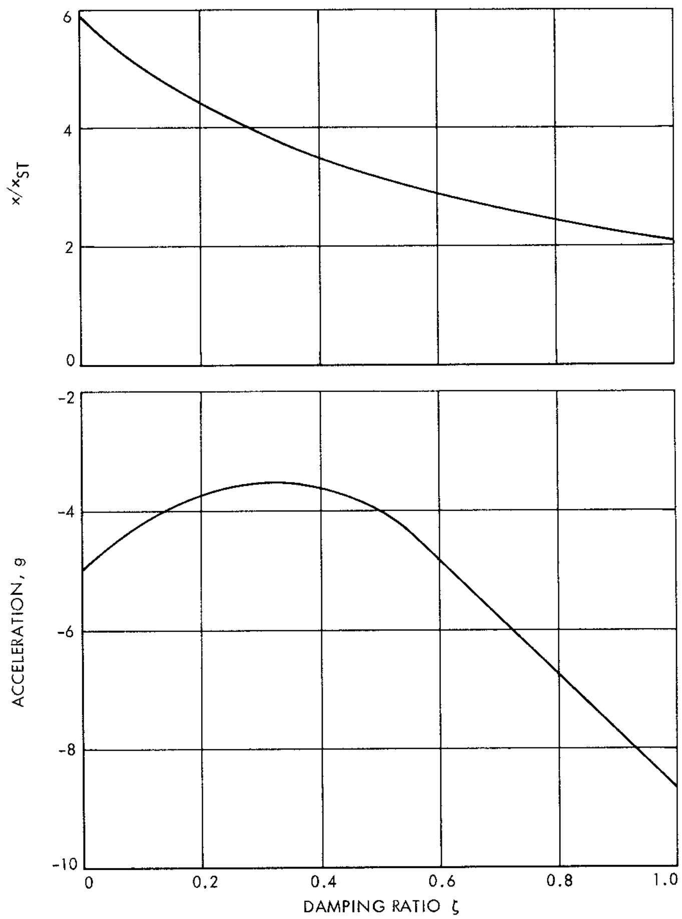

Fig. 6. Maximum acceleration and displacement versus damping ratio